

## **Моделирование механических свойств параболической антенны с композитным обтекателем в программной среде ANSYS**

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**Аннотация.** Актуальность работы связана с обеспечением надежности конструктивных элементов антенных постов радиолокационных станций при проектировании. С этой целью в среде ANSYS выполнено компьютерное конечно-элементное моделирование термонапряженного состояния алюминиевого параболического зеркала, закрепленного на кронштейне, закрытого композитным радиопрозрачным обтекателем, под действием собственного веса, ветрового воздействия и температуры. При моделировании слоистого композитного материала обтекателя использованы многослойные оболочечные элементы. Исследованы упругие деформации в зеркале при фиксированном воздействии ветровой нагрузки в диапазоне температур окружающей среды от  $-40$  до  $+50$  °C, в частности радиальные, осевые и нормальные перемещения. Установлено, что «развертывание» зеркала происходит при его охлаждении до  $-40$  °C на величину 1,2–1,3 мм, что сказывается на искажении амплитудно-фазового распределения поля в его раскрыве.

**Ключевые слова.** Радиолокационная станция, термонапряженное состояние, обтекатель, конечно-элементная сетка.

## **Modeling of Mechanical Properties of a Parabolic Antenna with Composite Radome in ANSYS Software Environment**

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**Abstract.** Modeling of thermostressed states of a parabolic aluminium mirror covered by radio transparent radome under gravity, wind and temperature effects is performed. A procedure is described to prepare a geometrical model for calculations in the ANSYS

software environment, and to create finite-element meshes. The calculated results are presented for the temperature range of  $-40^{\circ}\text{C}$  to  $+50^{\circ}\text{C}$ .

**Keywords.** Radar, thermostressed states, radome, finite-element meshes.

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## 1. Introduction

The most troubles and failures in radars refer to mechanics of power electric drives and structural elements of antenna posts in particular. Issues of radar structural strength influenced by mechanical loads appearing in the process of operation have been considered in papers [1–3]. Thermal strains developing in structural elements of radars due to nonuniform heating can result in additional errors in measurement of object coordinates [4–6].

Studies of various problems in mechanics being carried out at the engineering stage are generally performed with the use of analytical methods or experimental base. The idea of general behavior of structure under external influences is generated on the basis of analytical methods. Experimental base collected from similar structures, which however differ from a design project, allows the evaluation of structural behavior under various external factors. Despite a great number of studies performed both with the use of analytical and experimental methods, the problem of elimination of elastic strains arising in radar structural elements under various external influences and affecting their radio technical characteristics remains unsolved. The applied methods of analysis do not always meet the up-to-date requirements for the manufacturing accuracy of structural elements, or these methods cannot be applied at the design phase.

The solution of these problems considering all peculiarities of the radar mechanics with the required degree of accuracy at the design phase can effectively be performed with the use of finite-element (FE) modeling.

## 2. Statement of a Problem

### 2.1. Subject of Research

The basic elements of a typical radar antenna post are a parabolic aluminium mirror secured on a bracket, and a radome made of radio transparent composite material for reducing the external factors influencing the antenna system electrical performance (fig. 1). It is obvious that under the influence of wind, gravitation and temperature variations, the mirror reflective surface profile can distort what affects the radar accuracy characteristics.

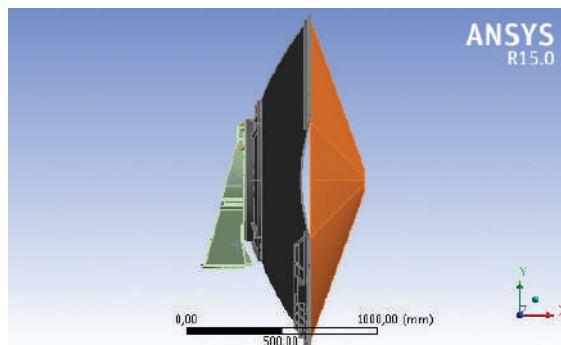


Fig. 1. Subject of research

Calculations of thermal strains have been carried out in the ANSYS Workbench software environment with the use of the following modules: ANSYS DesignModeler (preparation of CAD geometry for calculation, ANSYS Meshing (meshing for calculations by finite element method), ANSYS Composite PrepPost (setting of composite material properties), ANSYS Mechanical (statement and solution of problems in mechanics, analysis of results) [7].

## 2.2. Geometrical Model

As a rule, a computing engineer is given a design geometrical model developed in a CAD — modeling package of solid body properties. These models are highly to create accurate design documentation. However to prepare a FE model and carry out calculations, this elaboration seems to be unnecessary as it results in generation of a great number of small-sized finite elements. This increases the required computer calculations and does not improve calculation accuracy. Therefore, as a rule, the model is idealized and simplified. The model geometry is represented by shell bodies wherever possible. The thickness of a shell is assigned in accordance to the 3D-model of the structure. Considerable thickenings and massive blocks appeared in the geometry may be left in the form of solid bodies, i. e. bodies having a volume (fig. 2).

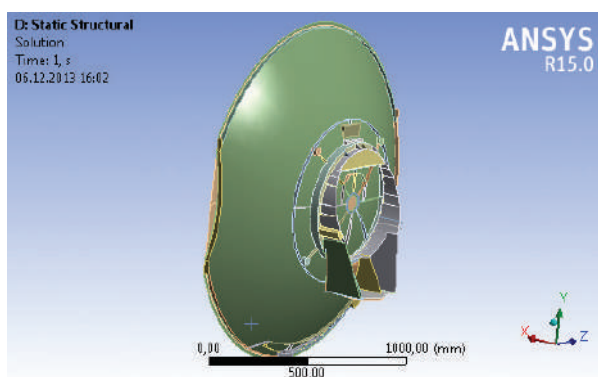


Fig. 2. Geometrical model of a problem

### 2.3. Finite-Element Model

In some places of geometrical model, the shells may be difficult to connect. Besides, it may turn out, that such places for connection are in great numbers. Packages in automatic tool in CAD to correct the geometry in such cases are known to the authors of this paper. An alternative variant of connecting geometrical objects is presented by the objects of a finite-element model. They are contact regions and mesh connections.

The formation of contact regions involves framing of some special elements which provide the deficiency of mutual penetration of surfaces. In the case under consideration, the only type of contact interaction that makes sense is a bonded connection. With this type of contact interaction, bonds are created between the surfaces, which do not allow them to move relative to each other in the space (fig. 3).

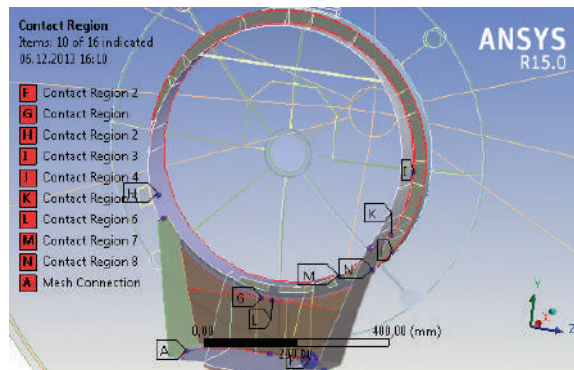


Fig. 3. Bonded connections in model

By default, the contact interaction is realized by means of penalty functions. The penetration of surfaces is possible. However, it is restricted by the force of contact springs. Nevertheless, this decision is practically equivalent to creating a rigid constraint of surfaces. If necessary, it is possible to eliminate the movement of surfaces completely by using formulation MPC (Multipoint Constraints, equations of constraints between sets of points) in settings of a contact region.

Unlike the contact regions, the Mesh Connections make no additional contact elements. The mesh is generated on the connectable surfaces separately, whereupon the joints, which fall on the Ribs, selected for connection, are contracted and merged. This results in some distortion of the model geometry, which may be allowable with small geometrical dimensions of gaps.

### 2.4. Setting of Radome Composite Properties

One of the objects under consideration in the problem is a radome made of composite material. The radome has certain rigidity and, when secured to the mirror, it exerts some effect on the mirror deformation under the in-

fluence of external loads. The properties of a composite material used in the radome (Young modulus, coefficient of thermal expansion) differ from those of aluminium. Besides, the composite material is anisotropic. Since the effective (average) mechanical properties of a laminated composite material are unknown, then the modeling by means of multilayer shell elements is used.

The mirror radome is a fiberglass honeycomb-cored sandwich-panel. As an example, reinforced plates (sandwich-panels) are made of monolayers based on glass cloth T-10–80, using special technology. The glass-honeycomb panels SSP-1 is used as a honeycomb core. The mechanical properties of materials taken for carrying out these calculations are given in Table 1.

Table 1

**Mechanical properties of composite materials**

Mechanical Properties		T-10–80	SSP-1
Young modulus	$E_x$ , MPa	27 000,0	1,0
	$E_y$ , MPa	17 500,0	1,0
	$E_z$ , MPa	17 500,0	255,0
Poisson ratio	$\nu_{xy}$	0,6	0,49
	$\nu_{zy}$	0,3	0,001
	$\nu_{xz}$	0,3	0,001
Modulus of shearing	$G_{xy}$ , MPa	8 437,5	$1,0 \cdot 10^{-6}$
	$G_{zy}$ , MPa	6 730,8	37,8
	$G_{xz}$ , MPa	6 730,8	70,8
Coefficient of linear thermal expansion	$\alpha_x$ , $^{\circ}\text{C}^{-1}$	$5,5 \cdot 10^{-6}$	—
	$\alpha_y$ , $^{\circ}\text{C}^{-1}$	$2,5 \cdot 10^{-5}$	—
	$\alpha_z$ , $^{\circ}\text{C}^{-1}$	$2,5 \cdot 10^{-5}$	—

The conjugation of layers in the mirror radome composite material as ideal, and the properties of the adhesive film between the honeycomb core and reinforcing plates as neglected are usually assumed while modeling. According to the manufacturing technique, the cloth is spread out over the cone surface to cover it in sectors. The sector width is  $30^{\circ}$ , and additionally, there is a 10 mm strip to provide overlapping with the next sector. One layer employs 12 sectors of this type. The reinforcing fibers in the layers are oriented differently: there are layers with fiber orientation “G”, “V”, and “F”. For layer “V”, the reinforcement axis direction is parallel to the bisectrix of the sector being covered, where the thickness of a monolayer is 0,25 mm. For layer

“G”, the reinforcement axis direction is perpendicular to the bisectrix of the sector being covered, where the thickness of a monolayer is 0,25 mm. Layer “F” is arranged in the form of a ribbon running along the radome flange (at a radius of more than 1594 mm) and consists of two monolayers of cloth T-10–80 with the perpendicular reinforcement axis direction: along and perpendicularly to the cone generatrix where the monolayer thickness is 0,5 mm.

### 2.5. External Influences and Kinematic Restrictions

Temperature variation is one of the external loads. A value of plus 22 °C is taken as a temperature required when performing the antenna post mechanical assembly. Therefore, any deviation of this temperature will cause thermal expansion (contraction) of the structural elements. Since the reflector is represented by a composite material, and the parabolic mirror, by aluminium, and moreover they are attached along the perimeter by screwed connections, elastic strains are expected to appear in the mirror, which result in distortion of its form.

In the antenna-to-isolated structure part attaching point, we “secure” all the displacements of antenna structure, attaining a fixed-ended connection (zone A in fig. 4). In this case, the bracket acts as a cantilever taking the loads from the mirror.

Apart from gravitation and constant temperature, the model is influenced by a load from wind pressure. The wind pressure may act from any side; however, the most hazardous event is when the full force vector direction coincides with the axis of cylinder. In this case, a peak torque appears in the fixed-ended connection. The wind pressure value is adopted according to the Russian third wind zone. The sum of the static and pulse components of the wind pressure equal to 600 Pa is assumed. To simplify the solution of the problem, the total pressure may be applied statically without solving the dynamical problem. Thus, a stress-strain state conservative estimation is performed.

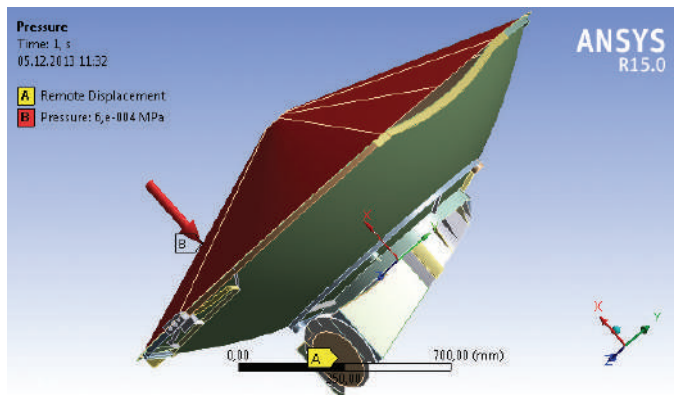


Fig. 4. Securing the structure in the bracket base and wind pressure

### 3. Analysis of Obtained Results

#### 3.1. Plotting of Normal Displacement Diagrams

Displacement of each point of the mirror surface is a sum of a mirror rigid-body displacement and a mirror deformation. The bracket strain may result in the mirror displacements which cause no distortion of the reflective surface. Since the result of the analysis is the mirror distortion under the influence of applied loads, then it is necessary to have a component extracted from the calculated displacements, which is conventionally called as “rigid”. Since the mirror paraboloid is a body of revolution, then, practically all the estimated results are convenient to consider in cylindrical coordinate system, the center of which coincides with the vertex of paraboloid.

For subsequent calculations, the radial coordinates of the mirror points in the earlier cylindrical coordinate system is used.

Compiling this data table will make it possible to considerably automate the plotting procedure for the basic diagrams of interest. Knowing the paraboloid equation in the cylindrical coordinate system, the angle between the normal and Z-axis for each point may be determined. Later on, this data table is used for determination of the displacement vector projection onto the normal surface.

When plotting the axial displacement diagram, it is necessary to take into account the rigid component determined from the displacements in the joint-connection. For different boundary conditions, the rigid component will be different. The obtained displacements as UV is designated (fig. 5, a). If the radial displacements in the cylindrical coordinate system is built, then there is a rigid displacement component along the global Y-axis (fig. 5, b).

It is necessary to take into account the earlier calculated displacements of the mirror as a rigid body, calculating the radial displacements in the cylindrical coordinate system connected with the center of parabolic mirror. It is necessary to perform a Y-shift for the loads under consideration. In subsequent research and temperature change, it is necessary to take into account the fact that the mirror rigid-body displacements will differ. The analysis is performed in the Cartesian coordinate system connected with the paraboloid vertex. Then, having the earlier obtained table of the radial coordinates for each joint, as well as the coordinates in the Cartesian system, the radial displacements without the rigid component of motion may be determined (fig. 6).

The need of calculations in the Cartesian coordinate system is determined by the fact that the rigid displacement component shall be deducted before the transformation of the displacement vectors into the cylindrical coordinate system.



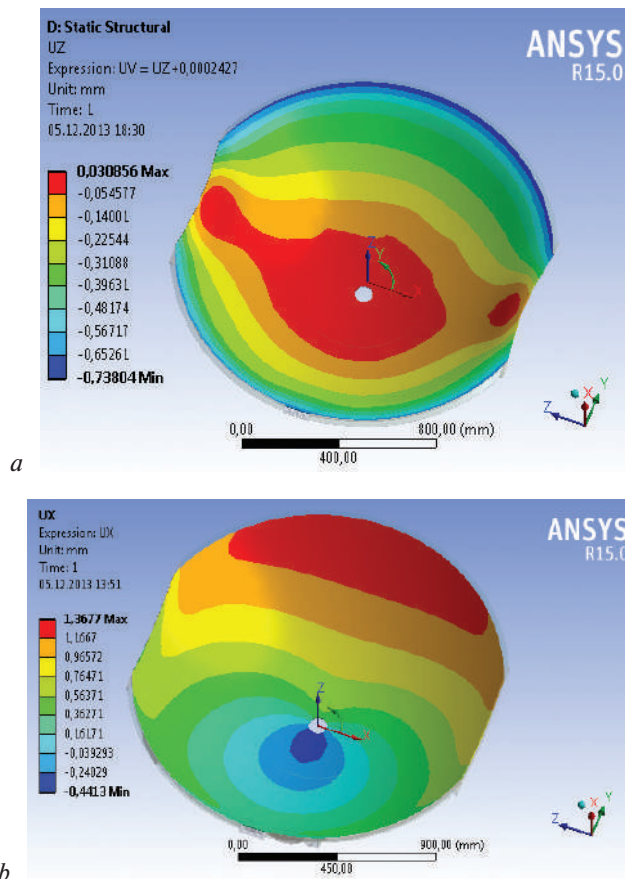


Fig. 5. Axial displacements minus rigid component (a);  
 radial displacements (full) (b)

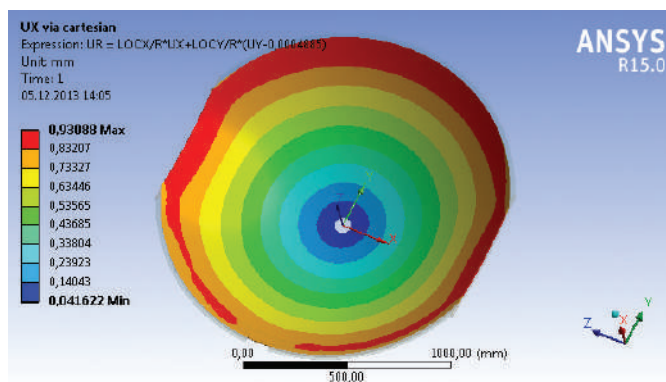


Fig. 6. Radial displacements minus rigid component



Upon compiling a table of radial and axial displacements in the cylindrical coordinate system with the table of angles between the normal and the axis of paraboloid being taken into account, the normal displacements of the mirror surface, which set conditions for its distortion may be determined.

### ***3.2. Study of temperature variation influencing the mirror surface distortion***

Let us perform a parametric study and plot maximal displacements being normal to the mirror surface versus the applied temperature load. The temperature step equal to 10 °C is assumed. It is necessary to trace the deformation behavior since not only the temperature loads are presented in the model. Therefore, let us plot normal displacement diagrams for given temperature values. The study of the calculation data shows that the deformation behavior at a temperature of 40 °C is practically the same as that at 50 °C (fig.7), with the only difference that the observed displacements differ in value. Maximal displacements arise along the mirror perimeter and they are positive, i. e. the mirror “curls up (contract)”.

The modeling has shown that at the temperature of plus 30 °C, the displacements deviate from the axially symmetric shape as far as the temperature strain value reduces with the temperature approaching to plus 22 °C. Increasingly more relative influence is exerted by gravitation and wind pressure. However, in bulk, their influence on the mirror deformation is insignificant, and therefore the maximal normal displacement value keeps on reducing. In the design temperature range, a value of plus 20 °C is the closest to the initial temperature of mechanical assembly. Thermal strains are practically absent, and the mirror distortion, caused by the other loads, is insignificant. The displacement diagram differs significantly in its behavior from the previous diagrams and represents a transition to a cooled state. When the temperature decreases from plus 10 °C, the behavior of the normal displacement distributions becomes increasingly more axially symmetric. In this case normal displacements, maximal in modulus, will arise along the perimeter, however, they are negative. In the cooled state, the mirror tends “uncurl (expand)”.

Fig. 8 illustrates a maximal mirror surface deflection (distortion) versus temperature. To evaluate a degree of the mirror surface distortion, the modulus of normal displacements are used.

The practically piecewise-linear function is obtained with a point of inflection at a plus 22 °C. In this case, the coefficient of linear expansion of material was supposed to be independent of temperature. To increase the calculation accuracy, it is necessary to introduce this characteristic into the calculation model, as a material property.

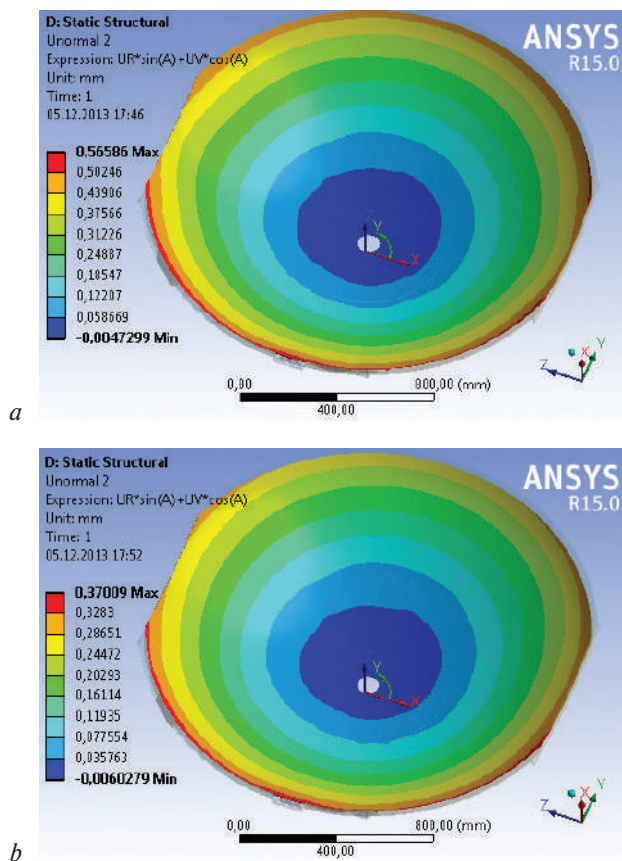


Fig. 7. Displacements in the normal to the mirror surface at plus 50 °C (a) and at plus 40 °C (b)

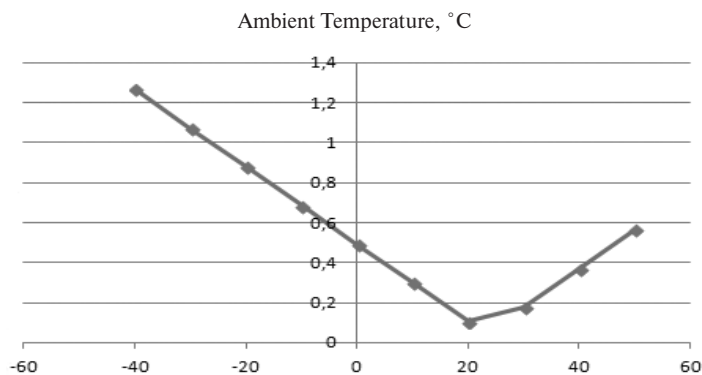


Fig. 8. Maximal normal displacements (mm) versus temperature

#### 4. Conclusion

The mirror expands by 1.2–1.3 mm, being cooled to a temperature of minus 40 °C, which leads to changing the amplitude-phase field distribution in the parabolic antenna aperture, and thereby to distortion of the antenna pattern. The evaluation of this distortion is a subject of further study. The ANSYS software makes it possible to perform quickly and accurately the modeling of thermostressed states of a parabolic aluminium mirror. Moreover, the automated preparation of the model geometry for calculations and the possibility of setting the composite material properties provide the consideration of all peculiarities of the designed structure.

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